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Article (Accepted Version)

Jablonka, Karolina Anna, Glovnea, Romeo and Bongaerts, Jeroen (2017) Quantitative measurements of film thickness in a radially loaded deep-groove ball bearing. *Tribology International*, 119. pp. 239-249. ISSN 0301-679X

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# Quantitative measurements of film thickness in a radially loaded deep-groove ball bearing

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Keywords: EHD lubrication, electrical capacitance, film thickness measurement, rolling element bearing

## 1. Abstract

The evaluation of the lubricant film thickness in machine elements working in elastohydrodynamic conditions is essential for the design aimed at improving their performance and durability. Among electrical methods for measuring the lubricant film thickness in these contacts the electrical capacitance is convenient to use because it relates directly to the film thickness by an inverse proportional relationship. Simultaneous measurements of optical film thickness and electrical capacitance have allowed the authors to perform quantitative evaluation of the film thickness in a model glass/steel contact, and develop a testing procedure which can be applied to steel/steel contacts. In the current paper a novel approach to film thickness measurements in rolling element bearings is presented. By replacing all but one steel ball with electrically insulating, ceramic balls evaluation of the lubricant film thickness in a radially loaded ball bearing has been achieved. The current procedure provides a valuable research tool for in-situ monitoring of lubrication condition, allowing studying the influence of operating parameters as well as the lubricant's chemistry.

## 2. Introduction

Electrical methods (electrical resistance, capacitance, inductance) have been widely used for the study of lubrication and lubricant film thickness measurements in model test devices as well as in machine components that work in elastohydrodynamic regime, such as rolling element bearings (Wilson [1], Leenders and Houpert [2], Magdun and Binder [3]), gears (MacConochie and Cameron [4]), cams (Vichard [5], Van Leeuwen et al. [6]), internal combustion engine piston ring (Hamilton and Moore [7], Sherrington and Smith [8]). When comparing the electrical methods, the advantage of electrical capacitance, over resistance, comes from the fact that according to the parallel-plate capacitor formula (Eq. 1), the lubricant film thickness can be easily extracted if the capacitance of the contact is measured:

$$C = \epsilon_0 \epsilon_r \frac{A}{h} \quad (1)$$

Here C is the capacitance between two parallel conducting plates of area, A, separated by a dielectric material with dielectric constant  $\epsilon_r$ , and thickness h.

Starting from the pioneering work on the lubrication of rollers by Crook [9], electrical capacitance has become the method of choice for the study of elastohydrodynamic (EHD) films between metallic surfaces. The disc machine, simulating a EHD contact between roller and raceway was also used by Dyson *et al.* [10] for the study of a wide range of fluids. Their work was then extended to grease

lubrication few years later Dyson, Wilson [11]. The authors showed the comparison of film thickness measured with electrical capacitance for greases and their corresponding base oils. Some of the first experiments involving electrical capacitance measurements on a rolling element bearing are due to Wilson [1]. He measured lubricant film thickness for a radially loaded double-row spherical roller and a single-row cylindrical roller bearing using the approach developed in earlier studies. Few years later Heemskerk *et al.* [12] presented the development of the “Lubcheck” apparatus designed for monitoring lubrication condition, based on electrical capacitance measurement. The bearing tested was a radially loaded deep-groove ball bearing lubricated with oil or grease. They were able to estimate the probability of asperity contacts, described in terms of metallic contact time fraction (PCT), and to measure the “lift-off” speed, defined as the speed at which the instrument recorded a PCT reading of 10%. The same instrument was then used by Leenders and Houpert [2], who measured the capacitance of radially loaded ball and spherical roller bearings lubricated with oil under the full-film and starved conditions, and then by Wikström and Jacobson [13] in their study concerning lubricant replenishment in spherical roller bearings. The “Lubcheck” instrument was also used to investigate the lubrication of bearings by refrigerant-lubricant mixtures Wardle *et al.* [14], Jacobson [15] and also to study the effect of surface micro-geometry on full film formation on a two-disc test rig Masen *et al.* [16]. Franke and Poll [17] presented a test rig in which, apart from the speed, temperature and friction torque the lubrication condition is also evaluated with the aid of the capacitance technique. In this work angular contact ball bearings were tested under pure axial load, lubricated with ten test greases. The same experimental setup and film thickness measurement approach was subsequently used by Gatzen *et al.* [18] and Wittek *et al.* [19] under grease lubrication.

One of the very limited comparative studies covering both, a model test device and full bearing tests, is due to Baly *et al.* [20]. The authors show the measurements of grease film thickness with optical interferometry on ball-on-flat rig, and electrical capacitance for a rolling element bearing. Experimental setup and procedure for the bearing part is similar to that used by Franke and Poll [17]. The study of lubrication in ball-on-flat contact has certain advantages, however there are few factors that play an important role in the lubrication of rolling element bearings, which cannot be replicated in model ball-on-flat devices. Longer times between successive over-rollings, lubricant flow inside a full bearing, the level of starvation, the contact area geometry, and the dynamics of the rollers and cage, all influence lubricant film thickness and are close to impossible to replicate in a ball and disc model contact (Lugt [21]). Also, a significantly different time scale of such experiments, in comparison with the full bearing tests, as pointed out by Wikström and Jacobson [13], is another disadvantage of ball-on-flat devices.

Murer *et al.* [22] used electrical capacitance sensors to measure the load distribution in rolling bearings. They successfully compare the experimental results with finite element analysis. Schnabel and co-workers [23] argue that the behaviour of contact capacitance in mixed regime is not well known thus they carried out an investigation into the running – in of rolling bearings by measuring EHL contact impedance. They conclude that further research is needed in order to understand the role of additives in impedance measurement.

Thus, despite the importance of optical film thickness measurements on model ball-on-flat rigs, there will always be need for full bearing tests, as only those can provide a full understanding of the behaviour of lubricants in highly loaded, bearing contacts.

Relatively recently Jablonka et al. [24] performed quantitative film thickness measurements with electrical capacitance in a ball-on-flat experimental setup. By using a chromium-coated glass disc simultaneous measurements of the lubricant film thickness by optical interferometry and electrical capacitance have been performed. A procedure to extract film thickness from the measured capacitance was developed and then applied to a steel-on-steel EHD contact. In the following paper [24] this approach was used to investigate the influence of lubricant's polarity on capacitance measurements and showed a surprisingly different behaviour of polar fluids in comparison to nonpolar. The current study is the extension of the previous work on EHD rig to the radially loaded deep-groove ball bearing. In the bearing tested all but one steel ball are replaced with ceramic balls, making it possible to follow the capacitance variation as the steel ball travels a full cycle. It should be noted that film thickness measurements with electrical capacitance on a radially loaded bearing are much more complex than for the axial load case, used in references [19, 20]. For axially loaded bearings the load is equally shared by all rolling elements making the calculations much simpler. In the measurements of radially loaded bearings, the load distribution within the bearing must be considered.

### **3. Experimental setup**

#### **3.1. Ball-on-flat device**

The first part of the experiments was carried out on a PCS Instruments EHD test rig. The rig has been modified in order to allow measurements of electrical capacitance between the disc and the ball. The disc and the ball are electrically insulated from each other and the rest of the rig and the capacitance is extracted from the impedance measured with a Solartron 1260 Impedance Analyser.

The measurements were performed at a frequency of 100 kHz and a signal voltage of 0.1 V was employed in most experiments. In the case of the mineral oil used it was necessary to modify testing (temperature) and impedance measurements (voltage) conditions in order to improve the quality of the data obtained. In the current settings the impedance is measured over one second and the average value of the capacitance is calculated and supplied directly by the instrument. For each speed 50 such readings are taken and the average value is used for the calculations. The experimental conditions are summarised in Table 1.

Two disc materials were used in this study. At first tests with a glass disc coated with chromium layer on the contacting surface were performed. This allowed simultaneous measurements of capacitance and film thickness with an optical interferometry method, as described in [24]. The chromium coated glass disc was then replaced with a steel disc and capacitance measurements have been carried out in similar working conditions of load and speed.

#### **3.2. Rolling element bearing setup**

The second part of the experimental work was performed in a bearing test rig. A 6306 ETN9 deep-groove ball bearing (DGBB) with a polymer cage and the inner and outer diameters of 30 and 72 mm was used. A radial load in the range between 1 and 6 kN was applied and the tests were performed over the range of entrainment speeds from 0.12 m/s up to 0.91 m/s (95–737 rpm). The Hertzian pressure, for the inner ring varied between 1.53 GPa and 2.64 GPa, while for the outer ring between 1.34 GPa and 2.27 GPa. The bearing was lubricated with a few drops of lubricating oil. In order to prevent starvation at high rolling speeds, small amount of oil was added before a sweep of speeds for each load. All tests were performed at room temperature and the bearing was run under self-induced

temperature conditions. The temperature was measured on the outer ring before and after a sweep of speeds for each load.

In the bearing under study all but one steel ball were replaced with ceramic balls of the same radius (made of silicon nitride –  $\text{Si}_3\text{N}_4$ ) and thus only the capacitance of the contact between the remaining steel ball and the raceways was measured. A “Lubcheck” capacitance voltage divider was used for the measurements. Details of this instrument and principle of operation can be found in [12, 13 and 25]. The values were displayed and recorded with an oscilloscope. The schematic of the experimental setup can be seen in Fig. 1.

Apart from the “Lubcheck” signal, the speed and load are recorded continuously, and also a pulse signal is registered when the steel ball passes the position of the maximum load at the bottom position of the bearing. This allows detecting any misalignment caused by improper bearing mounting, which could also lead to significant errors in film thickness calculation. The procedure allows following the steel ball as it goes around a full rotation of the inner ring and recording very rapid variations of the signal measured. For each speed setting five measurements were taken, each consisting of three or four cycles, depending on the speed. The “Lubcheck” voltage is inversely proportional to capacitance, meaning that low voltage corresponds to high capacitance and hence a thin lubricant film or a large surface area.

The “Lubcheck” signal on the left side of Fig. 2 corresponds to the position where the steel ball is around the centre of the unloaded zone (top position in Fig. 1). The ball then moves clockwise through the unloaded zone (flat region) and when it enters the loaded region, the voltage decreases (capacitance increases) as the load on the steel ball increases up to the maximum value at the point in the centre of the loaded zone (bottom position in Fig. 1). For this position minimum voltage reading (maximum capacitance) is recorded. From this point forward the load on the steel ball is gradually decreasing causing increased voltage (decreasing capacitance) up to the point when the steel ball leaves the loaded zone and the voltage signal remains almost constant. In Fig. 2 almost five full cycles are seen. The effect of the bearing alignment on the electrical voltage recorded by the “Lubcheck” instrument will show up as a “Lubcheck” voltage shape that does not match the contact area development.

#### **4. Experimental procedure**

##### **4.1. Lubricants tested**

The properties of the lubricants tested are listed in Table 2. Ball bearing tests were performed with a fully formulated mineral oil and a polyalphaolefin base oil (viscosity grade 48 according to ISO 3448). The viscosity was measured with an Anton Paar rotational viscometer, SVM 3000, while the pressure – viscosity coefficient was extracted from EHD film thickness measurements following a procedure detailed by Lafauntain [26]. For the ball-on-flat experimental setup, in order to extend the range of film thickness measured for mineral oil with the steel disc (limited by total capacitance value) it was necessary to increase the temperature up to 40°C. In the case of PAO base oil it was decided that instead of increasing testing temperature, a lower viscosity polyalphaolefin will be used and PAO4 was chosen.

#### 4.2. Dielectric constant evaluation

The dielectric constants of the lubricants at Hertzian contact pressure can be calculated from Clausius-Mossotti equation in the form given in Matveyev [27], if the density in the same conditions is known:

$$\frac{\varepsilon-1}{\varepsilon+2} = \frac{\alpha\rho}{3M} \quad (2)$$

where  $\varepsilon$  is the relative permittivity,  $\alpha$  is the polarizability,  $M$  is the mass of one molecule of the dielectric and  $\rho$  its density.

There is a number of models and vast amount of experimental data describing density-pressure relationship of lubricants with the formula proposed by Dowson and Higginson [28] most commonly used in lubrication studies. Fig. 3 shows the dielectric constant change with pressure calculated based on different density-pressure models. It can be seen that the Dowson-Higginson model tends to underestimate the lubricant's density, leading to lower dielectric constant, at higher pressures. Experimental data by Ståhl and Jacobson [29] show similar results as Tait's equation of state with constants according to Bair [30], and thus were used as a pressure-density model for the dielectric constant calculations in this study.

#### 4.3. Film thickness calculations

Theoretical central film thickness for the steel disc experiments in ball-on-flat setup, as well as for the ball bearing was calculated using the Hamrock-Dowson isothermal formula for a point contact [30]. Additionally, the film thickness decrease due to the inlet shear heating was included by applying the correction factor according to Gupta et al [32].

The detailed information regarding film thickness calculation from the measured capacitance for ball-on-flat setup can be found in [24]. The calculations for ball bearing are based on the same principles; however there are few points that must be highlighted here.

From electrical point of view the capacitance of the bearing consists of the capacitance between the steel ball and the raceways, the capacitance between inner and outer rings, and some background capacitance. The last two were evaluated using a bearing with all ceramic balls, and the obtained value was subtracted from all the measurement data. The capacitance between the steel ball and the raceways on its turn consists of the capacitance of the inner ring and the outer ring contacts, which are in series. Each of those contains the Hertzian contact capacitance and the capacitance of the region outside the contact, which are in parallel. In order to calculate lubricant film thickness several assumptions, similar to those proposed in [1, 2] were made:

- film thickness was calculated for the maximum load position (see below)
- the ratio between inner and outer ring Hertzian contact area and film thickness was taken from Hertz's theory and Hamrock-Dowson formula
- the capacitance of the region outside the contact on the inlet side was calculated up to the point where separation between the surfaces reaches nine times the central film thickness and fully flooded condition was assumed [33]. The capacitance on the sides of the contact was calculated assuming that oil fills the entire gap up to the shoulder of the rings. Finally the cavitation area at the exit of the contact. This was detailed in [24]. Appendix A shows how the contact capacitance was calculated.

The static distribution of the load for the modified bearing was carried out numerically following the analysis by Hamrock and Anderson [34]. However in the present case the difference in elastic moduli of steel and silicon nitride was taken into account. Details of this analysis are shown in Appendix B. It is also important to note that the bearing clearance strongly influences the load distribution in the bearing. The bearing under study has a C3 clearance which is in the range between 13 and 28  $\mu\text{m}$  (radial clearance). For the calculations the middle clearance value was used.

## **5. Results and Discussion**

### **5.1. Ball-on-flat setup**

The purpose of the ball-on-flat tests with fully formulated mineral oil and PAO base oil is to show, like in the previous study [24], that in a well-understood contact the capacitive method gives the results close to optical methods and also theoretical predictions.

The results of capacitance measured for both lubricants with chromium-coated glass and steel disc are shown in Fig. 4. The difference in measured capacitance is caused by a mixture of causes, mainly testing temperature, contact pressure with glass and steel disc and dielectric constant of the lubricant. All of these will be taken into account as described above. Film thickness values, for two lubricants, extracted from measured capacitance according to the procedure shown in [24], together with the film thickness measured by optical interferometry and are shown in Fig. 5 for chromium-coated glass disc. The film thickness from capacitance measurements, for the steel disc together with that calculated by Hamrock – Dowson equation is shown in Fig. 6. The comparison of film thickness measured with optical interferometry, or calculated from theory, with the values extracted from the measured capacitance shows a very good agreement for both lubricants. The dotted lines limiting  $\pm 10\%$  from optical/theoretical results are shown as guidance.

In Fig. 7 the ratio of measured capacitance and dielectric constant of the lubricant (calculated for the contact pressure with Clausius-Mossotti equation) is shown as a function of lubricant film thickness. It can be seen that both lubricants follow the same trend for a particular disc, however, it should be noted that this is only possible when both lubricants have low polarity. For high polarity fluids the deviation of the effective dielectric constant from theoretically calculated was observed in thin films [24].

Based on data shown in Fig. 7, it is also clear that the two disc materials show different trends. The explanation for this lies in different contact areas and also the contribution of contact capacitance in the total measured capacitance. For this reason the graphs similar to the ones shown in Fig. 4 can only be used to extract the film thickness for nonpolar lubricants tested only under the same load conditions and with the same disc materials.

### **5.2. Deep groove ball bearing (DGBB)**

The current setup, in which only capacitance of the contacts between one steel ball and the raceways are measured, allows following the steel ball as it travels the full rotation cycle. The capacitance distribution within the bearing measured with mineral oil for a range of loads (1-6kN) and single speed (0.41 m/s) is seen in Fig. 8. As the elastohydrodynamic film thickness is relatively insensitive to load change, variation of capacitance for different loads must come mainly from the change of the contact area and, in a lesser extent from the variation of the dielectric constant with pressure.

Based on the capacitance distribution throughout the bearing, the values corresponding to the position of maximum load, and the highest capacitance, were taken as this point should provide most valuable information about the lubrication condition.

The results of the measured capacitance at the highest load position for a ball bearing are shown in Fig. 9 and 10 for mineral oil and PAO VG48, respectively. As expected, a change of capacitance with load and speed can be seen. According to the parallel-plate capacitor formula and the data shown in Fig. 9 and 10, it is evident that the resolution of the method, and related to that the accuracy of film thickness calculation from the measured capacitance, strongly depends on speed and load. The higher the load (contact area) and lower the speed (film thickness), the bigger the change of capacitance for a certain film thickness change.

Because the two tested lubricants have significantly different viscosities (Table 2), to make a direct comparison possible, the ratio of the total measured capacitance and dielectric constant at contact pressure is displayed as a function of film thickness in Fig. 11. The inner ring film thickness was calculated following the procedure detailed in [24]. Similar conclusion as for the ball-on-flat case (Fig. 7) can be drawn. However here, instead of two different disc materials, variable load conditions are shown. The effect is the same, different trend resulting from different contact area and related to this different contribution of contact capacitance in total measured capacitance.

From the comparison of the “Lubcheck” voltage measured for mineral and PAO VG48 oils shown in Fig. 12, it is clear that for the same load and film thickness conditions significantly different behaviour is observed for the two lubricants. The signals from mineral oil tests show much more noise in the loaded region, than the corresponding results for polyalphaolefin. If the direct comparison for the same film thickness, such as that shown in Fig. 12, was not possible, it would be assumed that the “spikes” observed on the oscilloscope display result from asperity contacts, causing short-circuits. However, since both lubricants were tested on the same bearing, it becomes obvious that this cannot be the only reason, as the surface roughness was the same for both experiments. Thus the explanation must also come from oils’ chemistry, namely from the higher conductivity of the mineral oil due to the additives it contains.

According to the measurements by Harvey *et al.* [35] polyalphaolefin base oil (PAO6 in their case) exhibits, as expected, very low conductivity, because it is purely synthetic hydrocarbon base oil without heteroatom-containing species, which are found in mineral oils. Mineral base oils generally show higher conductivity than PAO, however, the level of conductivity depending on the API Group as it varies with refinement level. Harvey *et al.* also measured the conductivity of fully formulated and commercial oils which showed, as expected, much higher conductivity than the base oils.

It is therefore concluded that the “spikes” observed in “Lubcheck” signal in mineral oil tests are not only due to the roughness but are also caused by the current leakage through thin lubricating film because of increased conductivity (or lower resistivity) of the oil layer. The conclusion presented may seem obvious, however, if the origin of this behaviour is not fully understood and appreciated, significant errors when interpreting the data are likely to be obtained. The assumption that the noise is a result of metal-metal contact may lead to underestimation of lubrication condition in rolling element bearings. The same thing applies when only the average capacitance values are used for data analysis, because the presence of “spikes” artificially increases capacitance value, thinner than actual film would be deduced.



### 5.2.1. Film thickness comparison

Film thickness values calculated from the measured capacitance together with theoretical values for 6kN load are shown in Fig. 13 for the mineral oil. A decrease of capacitance film thickness below theoretical line is seen for both rings. This deviation is about 9% for the inner ring and 6% for the outer ring at low entrainment speed (thin films) and about 8% for both rings at larger speeds. In the absence of separate capacitance measurements for the inner and outer ring contacts, the partition of the measured capacitance attributed to each contact was done by the ratios of the theoretical areas as mentioned earlier. This accounts for the difference between the film thickness for the two rings seen in this figure although it can be appreciated that the geometry of the convergent region for the outer ring gives better conditions for lubricant entrainment, and thus thicker film. Since the film thickness is extracted from the combined capacitances between the ball and inner and outer rings the same trend for both contacts is observed. This is the case for all experiments, therefore from this point only the film thickness of the ball-inner ring contact will be shown for comparison.

The accuracy and uncertainty of the capacitance film thickness measurement is difficult to estimate due to the number of assumptions included in the calculations. First of all such analysis would be based on the fact that the theoretical value is the “real” film thickness in the bearing. This strongly depends on the actual temperature of the lubricant, which can be different than the temperature measured on the outer ring of the bearing. Another issue is the uncertainty coming from the film thickness calculation from measured capacitance. It should be noted that the maximum load value and the load distribution within the bearing strongly depends on the clearance used for the calculations, and that will affect the Hertzian contact area especially at low loads. In addition since it is not possible to know the actual film thickness distribution within the EHD contact it is necessary to assume the flat shape (by using the parallel-plate capacitor formula (Eq. 1)) and it is known that the contact has regions of thinner film which gives the lubricant film the well-known horse-shoe shape. Therefore, the dotted lines limiting  $\pm 15\%$  from theoretical predictions in the following figure are displayed only for guidance and are not a claim of the accuracy of the method. The comparison of film thickness extracted from the measured capacitance and calculated from Hamrock and Dowson formula, is shown in Fig. 14 for 2, 4 and 6kN radial load (a, b and c, respectively). It can be seen that the majority of the capacitance results is within those limits over the whole film thickness range. The general position of capacitance results relative to theoretical ones, varies slightly with the load. It is believed that the reason for this lies in contact area calculation, which is strongly dependent on the value of the load in the centre of the loaded zone. The influence of clearance and load calculation approach is the strongest for the lowest load (1kN) and the least relevant for the highest (6kN).

Additionally, slightly different trends can be seen in case of PAO VG48 and mineral oil. Capacitance results for the mineral oil show a gradual decrease below the theoretical film thickness line, which is not seen with PAO VG48 oil. It should be noted that the mentioned deviation occurs at much higher film thickness than the range studied with PAO VG48.

In order to explain this difference, an analysis on the inlet shear heating influence was carried on. The two lubricants studied differ significantly in viscosity and their viscosity-temperature characteristics. PAO has a higher viscosity index (Table 2) and thus will show smaller viscosity change than the mineral oil. The consequence of that is clearly seen in Fig. 15, where an inlet shear heating correction was applied to Hamrock and Dowson film thickness.

The temperature measurement after completing this sweep showed an increase of 1°C. However, since the temperature was only measured on the outer ring, the actual temperature of the lubricant

going through the EHD contact may be higher. In most of the work already published, for practical reasons, the bearing temperature was measured on the outer ring; however, as proved by Joshi *et al.* [36], the cage temperature may be more representative, as the cage response to change of operating conditions is much quicker.

In Fig. 15 film thickness calculation at the starting temperature of the test and at a temperature 2°C higher are shown. The correction to allow for the inlet shear heating was done with the relationship given in reference [30]. It is obvious that the influence of temperature, in this very small range, on theoretical film thickness corrected for inlet shear heating is much stronger for mineral oil, due to already mentioned lower viscosity index. It can be seen that for high speeds even such a small change of temperature as 2°C can cause significant decrease of film thickness for mineral oil, up to 20%.

Another possible explanation for the difference between measured and theoretical film thickness values is the deviation of the contact area from that was assumed for calculations, at thick film conditions. A change of film thickness shape/distribution over contact area can also be considered. If that happens, central film thickness calculated from theory may no longer be representative as the average film thickness, and a parallel-plate condition will no longer be fulfilled.

## 6. Conclusions/Summary

Although relatively easy from the point of view of the experimental setup, the evaluation of lubricant film thickness in rolling bearing contacts, from electrical capacitance measurements becomes more complex when it comes to extracting actual film thickness values from the experimental data. The difficulties arise firstly from separating the capacitance of the elastohydrodynamic contacts between the rolling elements and the raceways from the total capacitance measured and secondly from calculating the film thickness from the capacitance of the contact. In the present paper these difficulties were overcome by extending a procedure developed for a ball-on-flat arrangement to a modified ball bearing. In the ball-on-flat model contact simultaneous measurements of the film thickness by optical interferometry and capacitance allowed defining the steps which need to be taken to extract quantitative film thickness values from capacitance measurements. Ball bearing capacitance tests have been performed in a ball bearing in which all but one steel ball were replaced by ceramic balls thus allowing the isolation of only two contacts. This in turn made possible the evaluation of various factors such as lubricant composition, properties and supply, load, temperature, and entrainment speed. The approach with only one steel ball presented in the current paper gives much more clarity than full-bearing measurements providing a valuable research tool with many potential applications. It allows evaluating the effect of lubricants and additives, monitoring how the lubrication condition changes with time, and validating existing theoretical models, to name just a few. Overall good agreement was obtained for both lubricants tested over the range of speeds and loads. In this paper it was shown that obtaining a general procedure for extracting lubricant film thickness from capacitance measurement it is possible, and the factors influencing the accuracy of the capacitance measurement are underlined, making it easier to take them into account when evaluating the data.

## Acknowledgement

Miss Jablonka is grateful to SKF Engineering and Research Centre, Nieuwegein, The Netherlands for financial support for this research. All authors acknowledge Mr Jos Storken who set up the test rigs at ERC and helped with test work and Mr Henk Mol (also SKF ERC) who provided the idea for the single steel ball setup.

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## Appendix A

$$C_{bearing} = C_{steel\ ball} + C_{between\ rings} + C_{background}$$

$$\frac{1}{C_{steel\ ball}} = \frac{1}{C_{inner}} + \frac{1}{C_{outer}}$$

$$C_{inner} = C_{inner}^{contact} + C_{inner}^{outside}$$

$$C_{inner} = C_{contact}^{inner} + C_{outside}^{inner}$$

$$C_{inner}^{contact} = \varepsilon_0 \varepsilon_r^{p_i} \frac{A_{inner}}{h_{inner}}$$

$$h_{outer} = h' h_{inner}$$

$$A_{outer} = A' A_{inner}$$

$$\frac{1}{C_{steel\ ball}} = \frac{1}{\varepsilon_0 \varepsilon_r^{p_i} \frac{A_{inner}}{h_{inner}} + C_{inner}^{outside}} + \frac{1}{\varepsilon_0 \varepsilon_r^{p_o} \frac{A' A_{inner}}{h' h_{inner}} + C_{outer}^{outside}}$$

## Appendix B. Static load distribution

The distribution of the static load in radially loaded rolling bearings has been analysed by Hamrock and Anderson [ 32] among others. They use Hertzian theory and consider the elastic deformation of the loaded balls to extract a relationship between the load at a given angular position and the geometry of the bearing and the elastic properties of the bearing material. The load at a given angle  $\psi$  is then given by:

$$F_\psi = K_j(\delta \cos \psi - P_d/2)^j \quad (B1)$$

In this relationship  $K_j$  is the combined stiffness of the inner and outer ring contacts,  $\delta$  is the normal approach between the two raceways,  $P_d$  is the diametral clearance of the bearing, and  $j$  is an index which for ball bearings takes the value 1.5.  $K_j$  in its turn is given by:

$$K_\psi = \frac{1}{\left\{ \left[ \frac{1}{(K_j)_o} \right]^{2/3} + \left[ \frac{1}{(K_j)_i} \right]^{2/3} \right\}^{3/2}} \quad , \quad (B2)$$

with

$$K_j = \pi k E' \sqrt{2 \mathcal{E} R / 9 \mathcal{F}^3} \quad (B3)$$

The elliptic integrals of first kind  $\mathcal{F}$  and of the second kind  $\mathcal{E}$  are functions of the curvature ratio  $R_y/R_x$ .

$$\mathcal{F} = \frac{\pi}{2} + \left( \frac{\pi}{2} - 1 \right) \ln \left( \frac{R_y}{R_x} \right) \quad (B4)$$

$$\mathcal{E} = 1 + \left( \frac{\pi}{2} - 1 \right) \left( \frac{R_y}{R_x} \right)^{-1} \quad (B5)$$

Finally  $E'$  is the reduced elastic modulus. Looking back at relationships (B2) to (B5) it is seen that they depend on the geometry, and the elastic modulus of the balls and raceways, thus when all balls are made out of steel these relationships give identical parameters. In the arrangement of this study only one ball is made out of steel thus  $E'$  is different for this ball from all the other, which are made out of silicon nitride.

The numerical procedure for calculating the distribution of load is shown in the flow chart in Figure B1. The steel ball is located at angular position  $\psi$  and the angle between two adjacent balls is denoted by  $\theta$ .

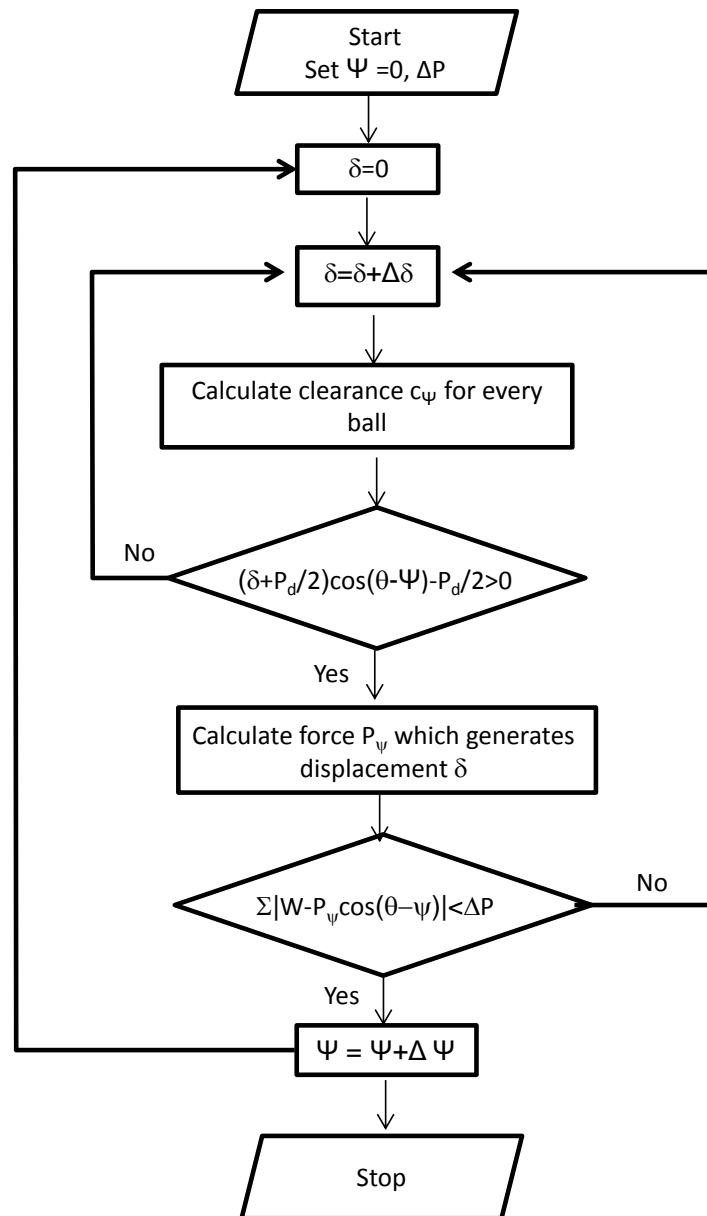


Figure B1. Flow chart for the calculation of static force distribution

Table 1 Testing conditions in ball-on-flat setup

Lubricant	Glass disc (0.65 GPa)	Steel disc (1.04 GPa)
Mineral oil	40N, 25°C, 0.10 V, 0.01-0.23 m/s	40N, 40°C, 0.05 V, 0.03-0.31 m/s
PAO4	40N, 25°C, 0.10 V, 0.01-0.46 m/s	40N, 25°C, 0.10 V, 0.01-0.46 m/s

Table 2 Lubricants' properties ( $\nu$  and  $\epsilon_r$  were measured)

	Mineral oil	PAO VG48	PAO4
Viscosity, $\nu$ , at 40°C (mm <sup>2</sup> /s)	99.0	46.4	17.2
Viscosity, $\nu$ , at 100°C (mm <sup>2</sup> /s)	11.4	7.9	3.9
Viscosity index	105	140	126
Pressure-viscosity coefficient GPa <sup>-1</sup>	24.5	16.5	15
Dielectric constant, $\epsilon_r$	2.3	2.1	2.1

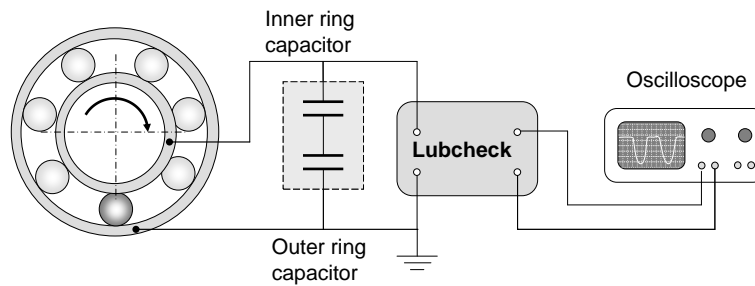


Fig. 1 Schematic of ball bearing experimental setup with single steel ball (dark) and remaining Si<sub>3</sub>N<sub>4</sub> balls



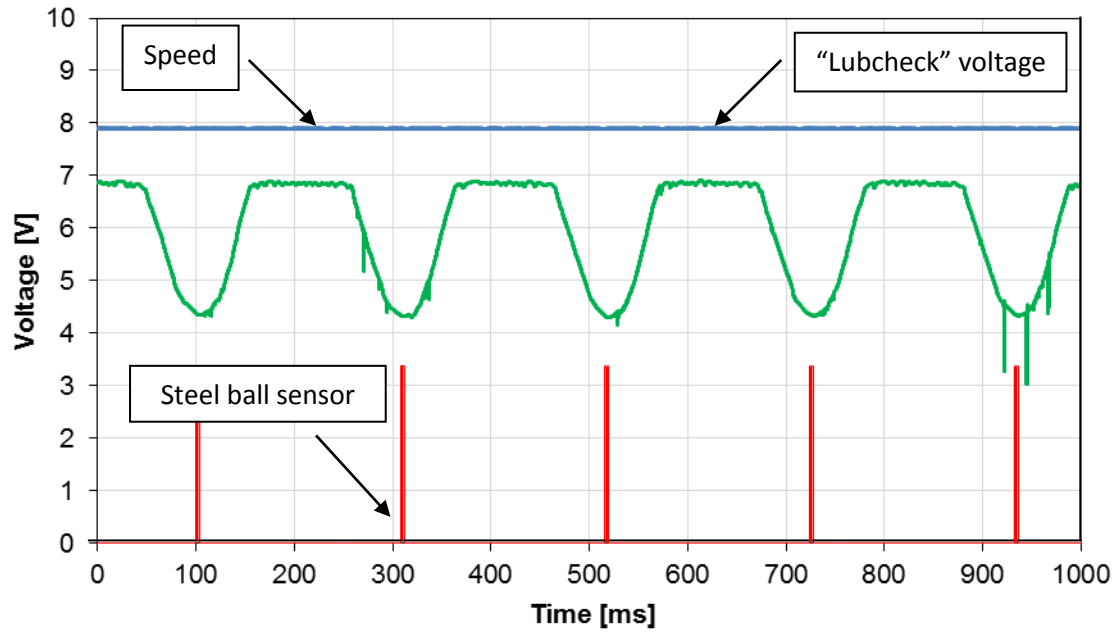


Fig. 2 Example of oscilloscope display data in DGBB tests (Mineral oil, 6kN, 0.91 m/s)

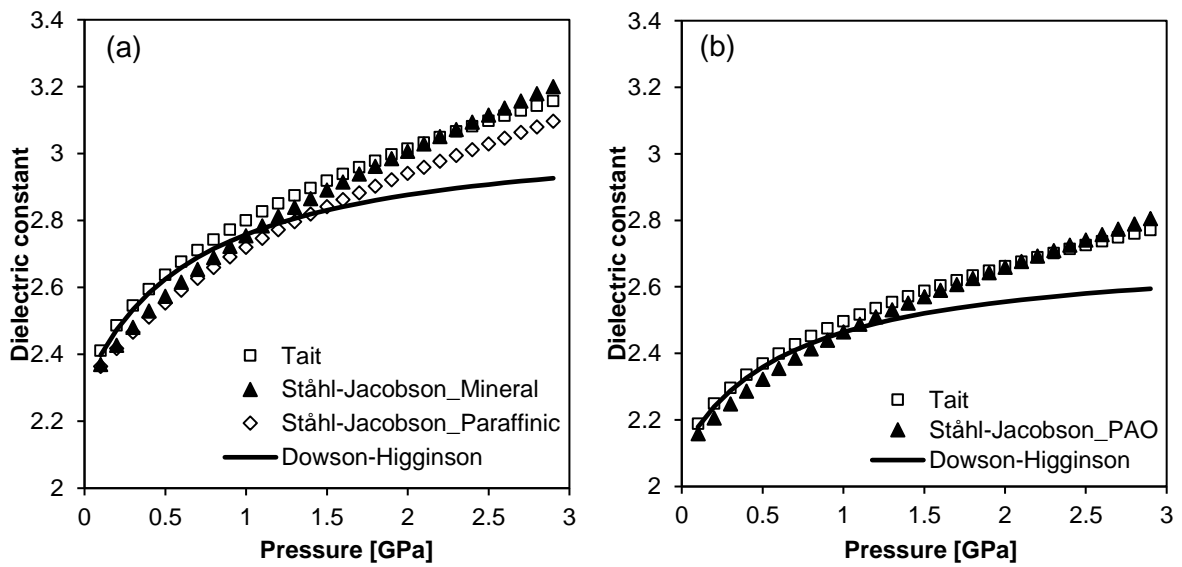


Fig. 3 Dielectric constant change with pressure based on different density-pressure models for mineral oil (a) and PAO VG48 oil (b)

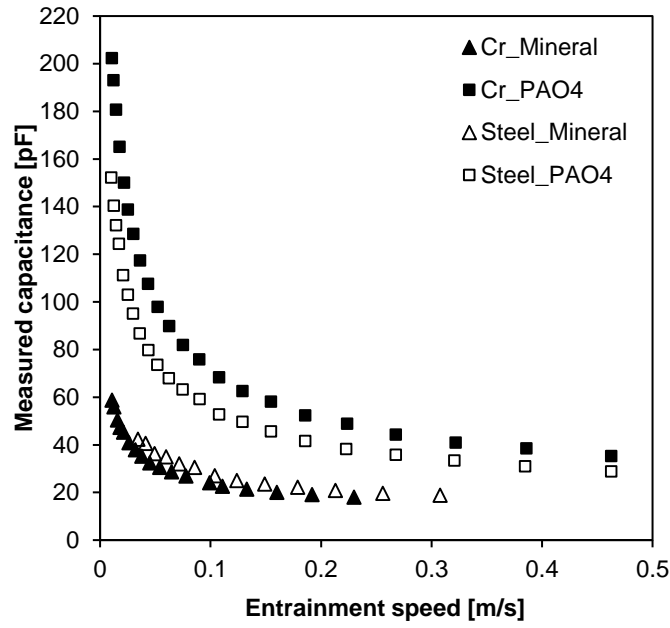


Fig. 4 Measured capacitance as a function of speed in ball-on-disc setup (Mineral oil and PAO4, Cr-coated glass and steel discs, 40N load)

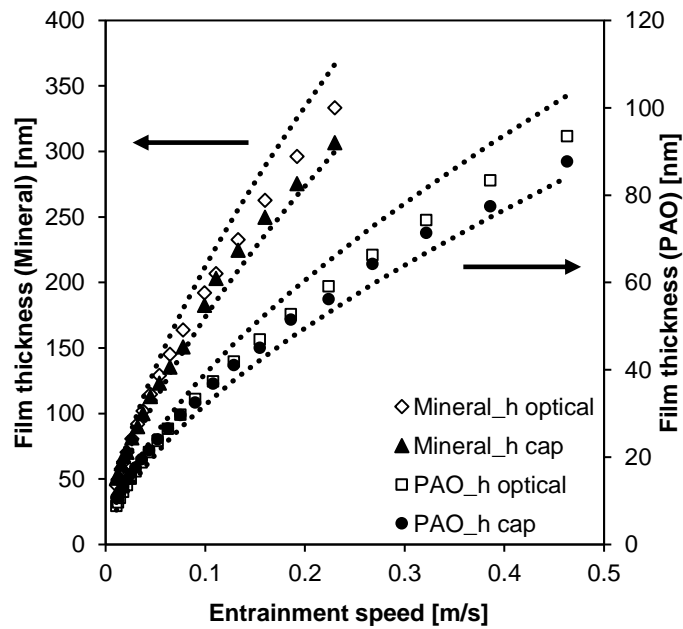


Fig. 5 Film thickness measured with optical method and capacitance in ball-on-disc setup (Mineral and PAO4 oil, 40N, Cr-coated glass disc, 25°C,  $\pm 10\%$   $h_{opt}$  dotted lines)

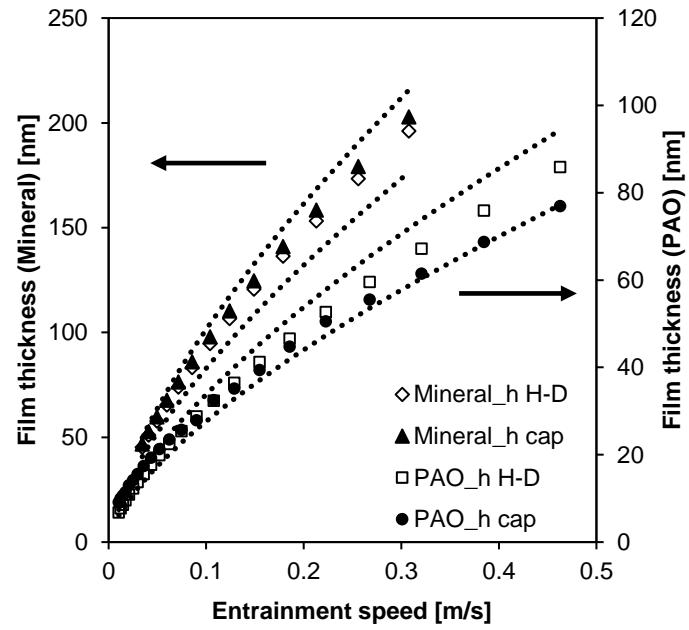


Fig. 6 Film thickness calculated from theory and measured with capacitance in ball-on-disc setup (Mineral and PAO4 oil, steel disc, 40N, 40°C and 25°C,  $\pm 10\% h_{theo}$  dotted lines)

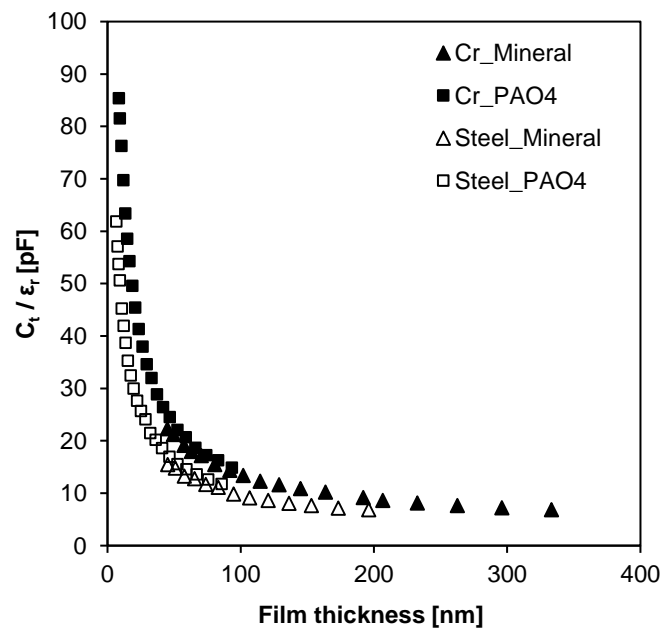


Fig. 7 Ratio of the measured capacitance and dielectric constant as a function of film thickness in ball-on-disc setup

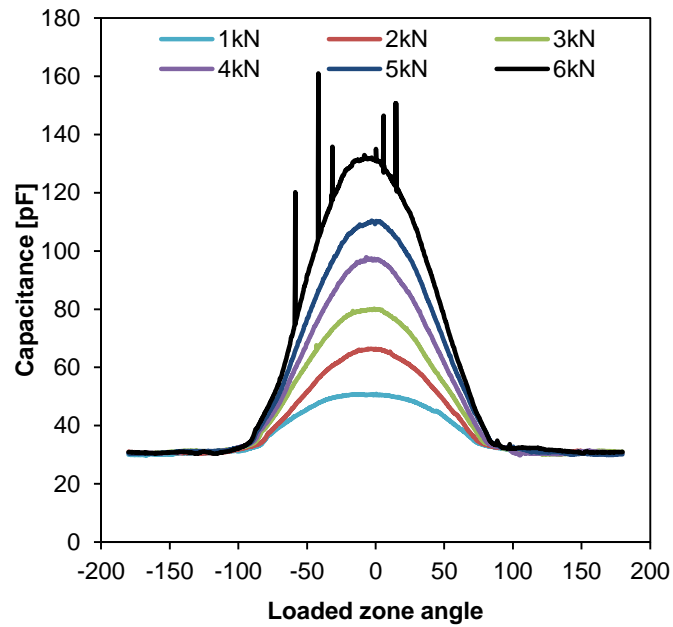


Fig. 8 Capacitance distribution over the unloaded and loaded zone in DGBB tests (Mineral oil, 0.41 m/s); Zero angle corresponds to maximum load position

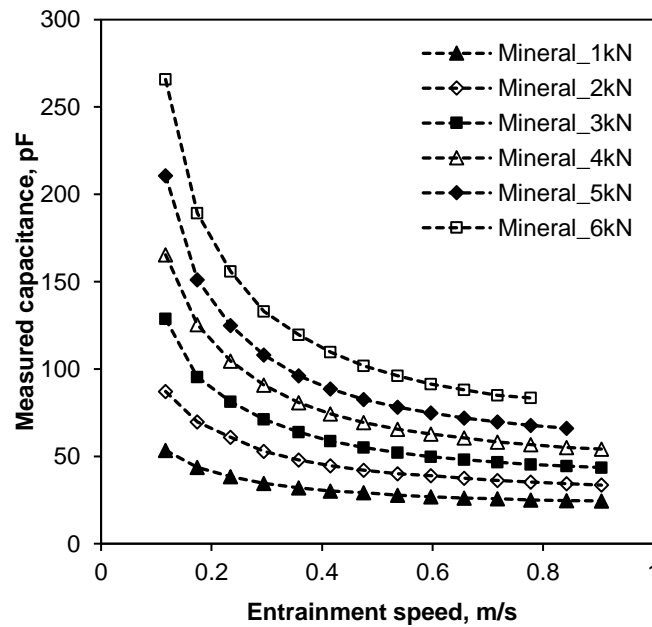


Fig. 9 Measured capacitance as a function of speed for mineral oil in DGBB tests

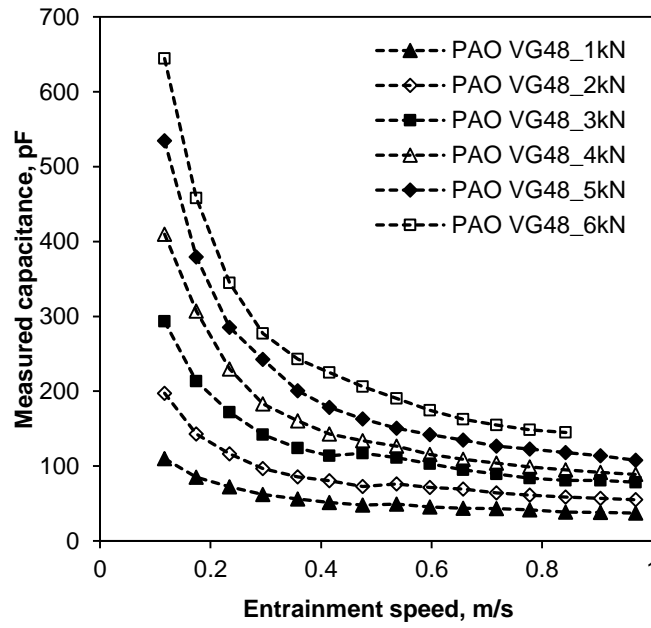


Fig. 10 Measured capacitance as a function of speed for PAO VG48 oil in DGBB tests

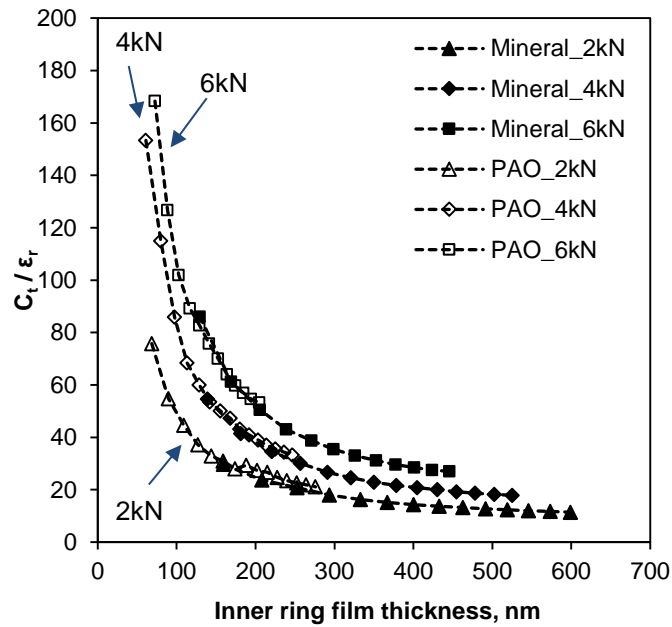


Fig. 11 Ratio of the measured capacitance and dielectric constant as a function of inner ring film thickness in DGBB tests

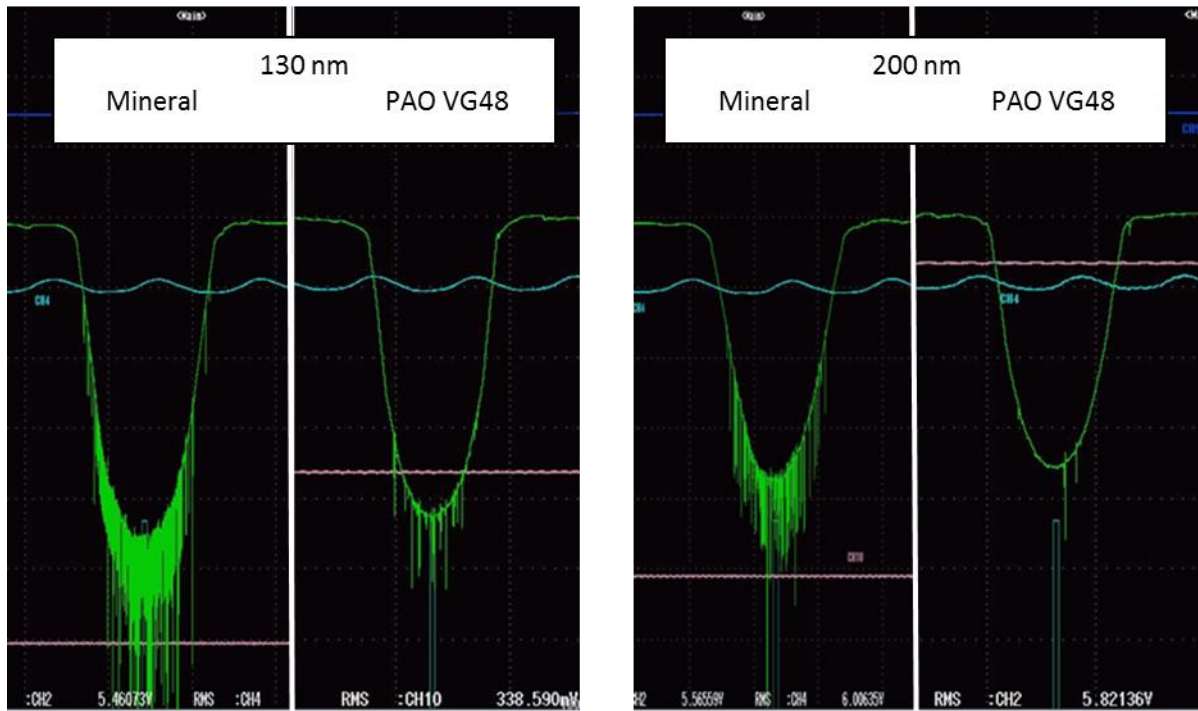


Fig. 12 Influence of additives on capacitance measurements in DGBB tests (inner ring film thickness, 6kN)

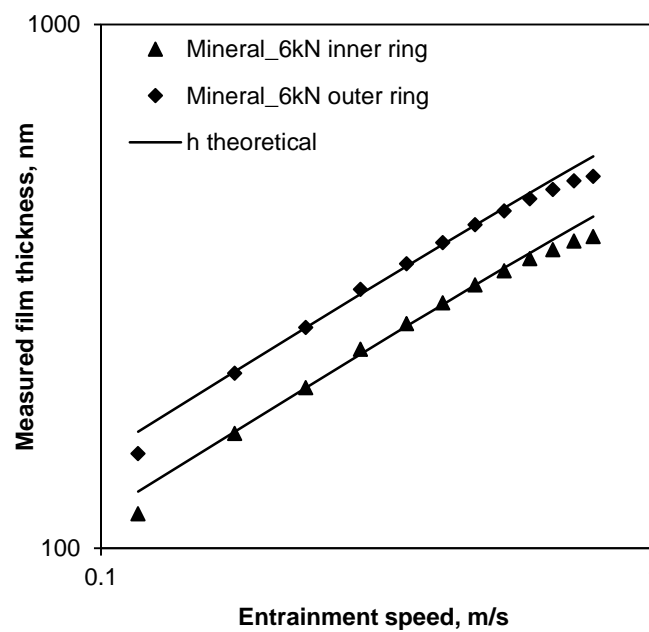


Fig. 13 Film thickness measured with capacitance compared with theoretical values for the inner and outer ring in DGBB tests (Mineral oil, 6kN)

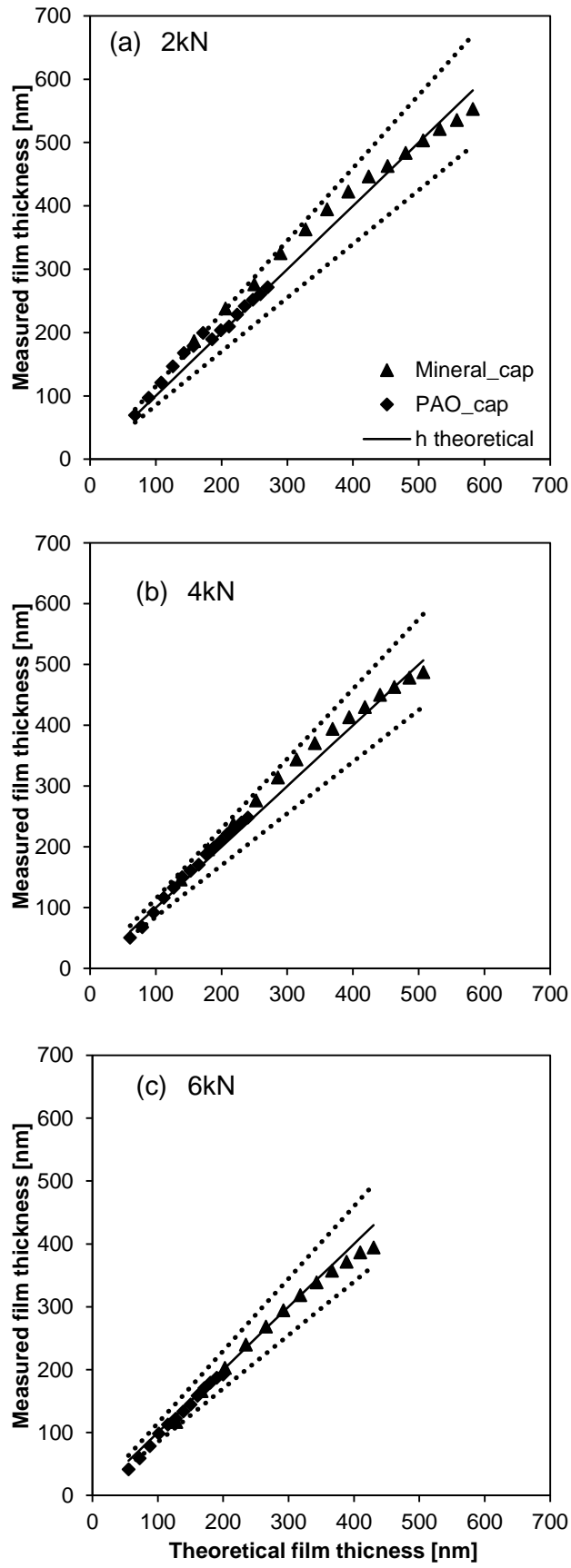


Fig. 14 Mineral and PAO VG48 oils in DGBB tests (inner ring film thickness,  $\pm 15\% h_{\text{theo}}$  dotted lines)

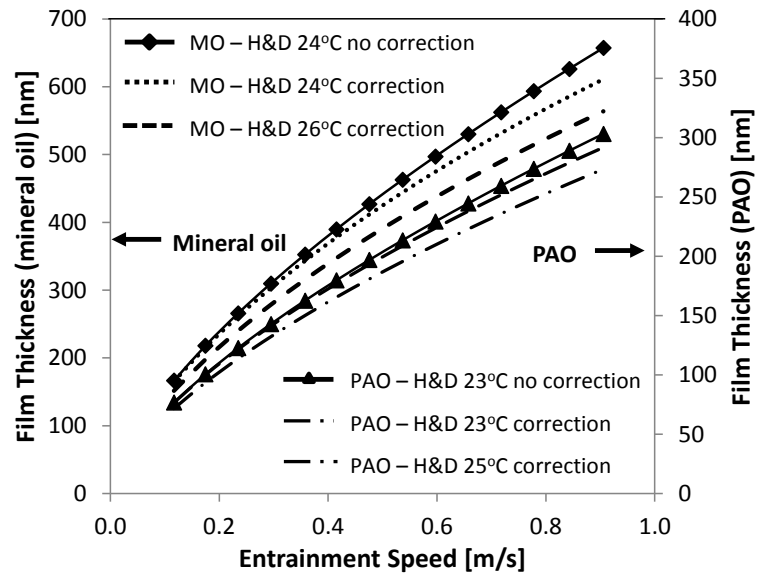


Fig. 15 Influence of the inlet shear heating correction on theoretical film thickness for mineral and PAO base oil in DGBB tests (1kN)